VARIABLE INNER VOLUME RATIO-TYPE INVERTER SCREW COMPRESSOR

TECHNICAL FIELD

The present invention relates to a variable inner volume ratio-mode inverter screw compressor in which an inner volume ratio, the ratio between suction capacity and discharge capacity of a screw compressor, is made variable.

BACKGROUND ART

There is conventionally provided a variable inner volume ratio-mode inverter screw compressor shown in Fig. 7 (for example, see JP 3159762 B2) as a variable inner volume ratio-mode inverter screw compressor in which an inner volume ratio thereof is variable.

According to this variable inner volume ratio-mode inverter screw compressor, when the inner volume ratio is required to be changed, a stepping motor 1 rotates a rod 2 to cause a variable VI (inner volume ratio) valve 3 to move backward, for example. At that time, a capacity controlling valve 4 moves backward together with the variable VI valve 3 moving backward, and when the variable VI valve 3 is fixed at a new set position, the capacity controlling valve 4 is fixed again on contact with the variable VI valve 3. Thus, a tip part of the capacity control valve 4 retreats to a position corresponding to the inner volume ratio after change to set an opening degree of a discharge port 5.

In this case, a pressure P_{d1} is detected which is generated immediately before the space which is formed by a rotor and the inner wall of a casing 7 during operation communicates with a discharge space, and the inner volume ratio is specified by giving a signal to a stepping motor 1 so as to minimize a difference ΔP between this detection pressure P_{d1} and discharge pressure P_{d2}. Otherwise, the inner volume ratio is specified by trend-analyzing parameters such as inlet pressure and discharge pressure during operation with a control device 10 to estimate optimum inner volume ratio and to give the stepping motor 1 a signal which indicates this optimum inner volume ratio.

In the structure described above, fluid, which is sucked from an suction port 6, is compressed by male and female rotors (not shown) in the casing 7 and is then discharged via the discharge port 5 to a discharge opening 8.

When the load applied to the variable inner volume ratio-mode inverter screw compressor varies in this situation so that capacity control is required, a hydraulic pressure piston 9 moves forward to make the capacity control valve 4 advance as far as a required distance based on the control command.

Therefore, a gap is generated between the variable VI valve 3 and the capacity control valve 4. The fluid on the way of its compression is bypassed to the suction side through the gap between the variable VI valve 3 and the capacity control valve 4.

However, the conventional variable inner volume ratio-mode inverter screw compressor disclosed in JP 3159762 B2 has the following problems.

That is, the variable inner volume ratio technology for the conventional variable inner volume ratio-mode inverter screw compressor makes the inner volume ratio of the compressed gas discharged from the discharge port 5 variable to achieve the highest compressor efficiency according to the high/low pressure conditions during operation, but the setting thereof corresponds to a full-load capability condition (100% load). It is disadvantageous in that the conventional variable inner volume ratio-mode inverter screw compressor is inefficient since capability regulation (unload control) is performed under a partial load capability condition (part-load) by bypassing the fluid on the way of its compression to the suction side through the gap between the variable VI valve 3 and the capacity control valve 4.

It is also disadvantageous in that the valve control mechanism of the conventional variable inner volume ratio-mode inverter screw compressor is complex since the variable VI valve 3 for changing inner volume ratio and the capacity control valve 4 for controlling capacity are provided so that the controlling mechanism of the variable VI valve 3 for the

time when inner volume ratio is changed and the controlling mechanism of the capacity control valve 4 for the time when capacity is controlled are required to be provided separately.

DISCLOSURE OF THE INVENTION

Accordingly, it is an object of the present invention to provide a variable inner volume ratio-mode inverter screw compressor which constantly achieves the highest efficient operation corresponding to a load (operating condition).

In order to achieve the above object, a variable inner volume ratio-mode inverter screw compressor of the present invention comprises a variable inner volume ratio valve changing completion time of a compression step in a screw compression section to make inner volume ratio variable;

an electric motor rotationally driving the screw compression section; and

an inverter controlling rotational frequency of the electric motor corresponding to a load.

According to the above structure, when compression capability is regulated corresponding to a load, the rotational frequency of the electric motor is controlled by the inverter. Capability is thus regulated without performing unload control. And the opening degree of the variable inner volume ratio valve is controlled to set the completion time of a compression step in the screw compression section to achieve the highest compressor efficiency according to the regulated rotational frequency of the electric motor. As a result, the highest efficient operation is constantly achieved corresponding to a load.

A variable inner volume ratio-mode inverter screw compressor of the present invention comprises a control section controlling an opening degree of the variable inner volume ratio valve based on suction side pressure and discharge side pressure of the screw compression section and rotational frequency of the electric motor.

According to the above structure, when the inner volume ratio is variable, the

opening degree of the variable inner volume ratio valve is controlled by the control section based on the suction side pressure and the discharge side pressure in the screw compression section and the rotational frequency of the electric motor. Therefore, the inner volume ratio is accurately and easily controlled so that the inner volume ratio achieves the highest compressor efficiency corresponding to the rotational frequency of the electric motor adjusted by the inverter by using a prescribed relation of a compression ratio, the rotational frequency of the electric motor and optimum inner volume ratio.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1A and Fig. 1B are structural diagrams showing a substantial part of a variable inner volume ratio-mode inverter screw compressor of the present invention;

Fig. 2 is a view showing a capability/inner volume ratio control system of the variable inner volume ratio-mode inverter screw compressor shown in Fig. 1;

Fig. 3 is a view showing a capability/inner volume ratio control system which is different from the capability/inner volume ratio control system of Fig. 2;

Fig. 4 is a view showing a relation of compression ratio and optimum inner volume ratio at each operation frequency;

Fig. 5 is a view showing the relation of freezing capability and compressor efficiency at each compression ratio;

Fig. 6A and Fig. 6B are views showing the relation of inner volume and pressure of a screw-type compressor; and

Fig. 7 is a cross-sectional view of a conventional variable inner volume ratio-mode inverter screw compressor.

BEST MODE FOR CARRYING OUT THE INVENTION

Hereinafter, this invention will be described in detail by way of preferred embodiments shown in the accompanying drawings. Fig. 1 is a schematic structural diagram showing a variable inner volume ratio-mode inverter screw compressor of this

embodiment. It is to be noted that Fig. 1A shows the case when inner volume ratio is low and Fig. 1B shows the case when inner volume ratio is high.

In Figs. 1A and 1B, reference numeral 11 denotes an electric motor, which has a stator 12 fixed on a casing (not shown) and a rotor 13 rotating and fixed on one edge side of a main shaft 14. The electric motor 11 is driven by an inverter 15. Both ends of the main shaft 14 are supported by bearings 16 and 17, and a screw rotor 18 is fixed on the other edge side of the main shaft 14. When the main shaft 14 is rotated by the electric motor 11, the screw rotor 18 rotates and suction gas is compressed by the screw groove (not shown) on the outer circumferential surface of the screw rotor. There is provided a discharge port 20 of a prescribed length in an axial direction, a cylindrical slide surface of the screw rotor 18, and the gas compressed by the screw rotor 18 is discharged from the discharge port 20.

One end of a plurality of rods 22, which are slidably supported by a supporting plate 21, is fixed on an end face of the slide valve 19 opposite the electric motor 11 side. The other end of the respective rods 22 is fixed on one coupling plate 23. A cylinder 24 is arranged at the center of the surface of the supporting plate 21 opposite the screw rotor 18 side, and the coupling plate 23 is fixed on a tip part of a piston rod 26 which is fixed on a side opposite to the screw rotor 18 side of a piston 25 contained in the cylinder 24. The slide valve 19 thus moves in an axial direction through the piston rod 26, the coupling plate 23 and the rod 22 as the piston 25 moves in an axial direction.

Working fluid, which is supplied to operating chambers located at both sides of the piston 25 in the cylinder 24, is controlled by a fluid control device 28 based on a control signal from a compression section controller 27. It is to be noted that the specific structure of the fluid control device 28 is not limited provided that the fluid control device 28 is constituted such that the piston 25 moves toward the screw rotor 18 as shown in Fig. 1A when inner volume ratio is reduced, whereas the piston 25 moves apart from the screw rotor 18 as shown in Fig. 1B when inner volume ratio is raised.

In the variable inner volume ratio-mode inverter screw compressor as constituted above, capability regulation to a load is performed by a revolution number control of the electric motor 11 by an inverter 15. This eliminates unload control when the capability is regulated, and preventing operational efficiency from lowering. Furthermore, a capacity control valve for capacity control is also eliminated, simplifying a valve control mechanism.

Meanwhile, the position of the slide valve 19 for the variable inner volume ratio is controlled by the compression section controller 27 so as to achieve the highest compressor efficiency corresponding to operating condition. When a low inner volume ratio command is issued, compressed gas is quickly discharged by moving the slide valve 19 (that is, a starting position of the discharge port 20) in an axial direction toward the electric motor 11 to advance the completion time of a compression step in the compression section. On the other hand, when a high inner volume ratio command is issued, compressed gas is slowly discharged by moving the slide valve 19 (that is, the starting location of the discharge port 20) in an axial direction toward the piston 25 to delay the completion time of a compression step in the compression section. That is, according to this embodiment, the variable inner volume ratio valve is composed of the slide valve 19.

As described above, when the revolution number of the electric motor 11 is set by the inverter 15 and the position of the slide valve 19 is set by the compression section controller 27, the suction gas sucked from a suction port is introduced to the screw rotor 18 through the electric motor 11. The suction gas is then compressed by the screw groove formed on the outer circumferential surface of the screw rotor 18 to be discharged from the discharge port 20 of the slide valve 19.

A description will hereinafter be given for the revolution number control of the electric motor 11 and the position control of the slide valve 19 according to this embodiment.

Fig. 2 is a view showing a capability/inner volume ratio control system of this

variable inner volume ratio-mode inverter screw compressor. In Fig. 2, a description is given for a screw compressor 31 mounted on a refrigerator, for heating coolant by compression as an example.

The refrigerator is composed of a screw compressor 31, a condenser 32, an expansion valve 33 and an evaporator 34, which are annularly connected in order. The high temperature-high pressure coolant discharged from the screw compressor 31 is condensed by heat exchange with cold water or air in the condenser 32, and becomes low temperature-high pressure liquid coolant to be supplied to the expansion valve 33. The low temperature-low pressure liquid coolant reduced in pressure in the expansion valve 33 evaporates by heat exchange with water in the evaporator 34 and becomes low pressure gas to return to the screw compressor 31. The cold water cooled in the evaporator 34 is used for cooling.

A temperature sensor 35 is mounted on the coolant pipe of the evaporator 34, and a detection signal, which indicates cold-water temperature Tw from the temperature sensor 35, is inputted into a revolution number output section 37 of a control device 36. The revolution number output section 37 then calculates a rotational frequency Hz of the electric motor 11 to obtain freezing capability required based on the difference between the inputted detection signal i.e. cold water temperature Tw as load side information and a preset temperature for example, and outputs the result to an optimum inner volume ratio output section 38 of the control device 36 and the inverter 15. The inverter 15 controls the revolution number of the electric motor 11 based on the received rotational frequency Hz. Regulating capability to a load is thus performed.

On the other hand, a low-pressure side pressure sensor 40 is mounted on the suction side of the screw compression section 39 which includes the screw rotor 18 and the slide valve 19, and a high-pressure side pressure sensor 41 is mounted on the discharge side thereof. The detection signal for indicating a low pressure LP from the low-pressure side pressure

sensor 40 and the detection signal for indicating a high pressure HP from the high-pressure side pressure sensor 41 are then inputted into the optimum inner volume ratio output section 38. The optimum inner volume ratio output section 38 detects the operating condition after setting the revolution number of the electric motor 11, based on the low pressure LP on the suction side and the high pressure HP on the discharge side which are based on the inputted detection signals. The optimum inner volume ratio output section 38 then performs arithmetic processing based on the low pressure LP, the high pressure HP and the rotational frequency Hz from the revolution number output section 37 and calculates the optimum inner volume ratio in the present rotational frequency Hz to output the result to the compression section controller 27. Consequently, the compression section controller 27 controls the operation of the fluid control device 28 based on the received inner volume ratio. The inner volume ratio control is thus performed corresponding to operating condition.

When the structure of the fluid control device 28 has an element which operates in proportion to the movement of the slide valve 19 in an axial direction (an external driving motor for operating a pilot valve and the like), the position of the slide valve 19 can be detected based on the operation position of the element. In this case, the detection signal indicating the position SV of the slide valve 19 and coming from the fluid control device 28 is inputted into the optimum inner volume ratio output section 38 directly or alternatively through the compression section controller 27. The optimum inner volume ratio output section 38 obtains the present inner volume ratio value based on the received position SV of the slide valve 19 to feedback control the optimum inner volume ratio value. This allows the variable inner volume ratio to be controlled accurately.

It is to be noted that when the structure of the fluid control device 28 (the structure being composed of piping and an electromagnetic valve, for example) cannot detect the position of the slide valve 19, the optimum inner volume ratio output section 38 cumulates

output inner volume ratio values from startup. Feedback control is performed by taking the cumulated inner volume ratio value as the present inner volume ratio value to calculate a controlled variable ΔVI to optimum inner volume ratio value.

Fig. 3 is a view showing a capability/inner volume ratio control system which is different from the capability/inner volume ratio control system of Fig. 2. Also in Fig. 3, the screw compressor 31 is mounted on the refrigerator. A control device 51 and an inverter 54 are constituted differently from the control device and the inverter of Fig. 2. Hereinafter, a description will be given mainly for the operations of the control device 51 and the inverter 54, and the members identical to the members shown in Fig. 2 have the same reference numbers as those in Fig. 2.

Similarly to the case of Fig. 2, the detection signal from a temperature sensor 35 which indicates cold-water temperature Tw is inputted into a revolution number output section 52 of the control device 51. The detection signal from the low-pressure side pressure sensor 40 which indicates the low pressure LP and the detection signal from the high-pressure side pressure sensor 41 which indicates the high pressure HP are inputted into an optimum inner volume ratio output section 53 of the control device 51. The revolution number output section 52 then calculates the rotational frequency Hz of the electric motor 11 to obtain required freezing capability based on cold-water temperature Tw, and the revolution number of the electric motor 11 is controlled by the inverter 54. Regulating capability to a load is thus performed.

The inverter 54 of this embodiment can detect the driving voltage V and driving current A (or driving power W) of the electric motor 11, and sends these detected driving voltage V and driving current A (or driving power W) back to the revolution number output section 52. The calculated rotational frequency Hz and the received driving voltage V and driving current A (or driving power W) are transmitted to the optimum inner volume ratio output section 53 by the revolution number output section 52.

Then, the optimum inner volume ratio output section 53, similarly to the case of Fig. 2, performs arithmetic processing based on the low pressure LP and high pressure HP from the pressure sensors 40 and 41, the rotational frequency Hz from the revolution number output section 52 and the position SV of the slide valve 19 from the fluid control device 28, to calculate the controlled variable ΔVI to optimum inner volume ratio and to output the result to the compression section controller 27.

Furthermore in this embodiment, the change transition of the driving voltage V and driving current A (or driving power W) from the revolution number output section 52 are stored in the optimum inner volume ratio output section 53. And the inner volume ratio is controlled so that the driving voltage V and driving current A (or driving power W) are minimized as the inner volume ratio control is repeated.

Afterward, similarly to the case of Fig. 2, the operation of the fluid control device 28 is controlled by the compression section controller 27 based on the received controlled variable ΔVI to feedback control the inner volume ratio corresponding to operating condition.

It is to be noted that in this case, similarly to the case of Fig. 2, when the structure of the fluid control device 28 cannot detect the position of the slide valve 19, the optimum inner volume ratio output section 53 calculates the controlled variable ΔVI of the optimum inner volume ratio value by taking a cumulated inner volume ratio value obtained by cumulating output inner volume ratio values from startup as the present inner volume ratio value.

In the optimum inner volume ratio output sections 38 and 53 of the control devices 36 and 51 shown in Fig. 2 and Fig. 3, arithmetic processing is performed to calculate controlled variable ΔVI to optimum inner volume ratio. However, the low pressure LP from the low-pressure side pressure sensor 40, the high pressure Hp from the high-pressure side pressure sensor 41 and the rotational frequencies Hz from the revolution number output sections 37 and 52 are stored in a memory in order. It is also possible that low pressure LP,

high pressure HP and rotational frequency Hz are compared with the low pressure LP, high pressure HP and rotational frequency Hz in the previous inner volume ratio operation so as to obtain controlled variable ΔVI to optimum inner volume ratio based on the change transition.

Fig. 4 shows the relation of the compression ratio indicated by the ratio (HP/LP) of the high pressure HP from the high-pressure side pressure sensor 41 to the low pressure LP from the low-pressure side pressure sensor 40 and the optimum inner volume ratio at each operation frequency Hz (=30 Hz, 60 Hz, 90 Hz). In Fig. 4, a straight line shows a theoretical value of the optimum inner volume ratio indicated by $VI=(HP/LP)^{1/k}$ (k: coolant specific heat ratio). This relation of the compression ratio, the optimum inner volume ratio and the operation frequency Hz is obtained at each coolant to include the relation to an arithmetic expression used when arithmetic processing is performed by the optimum inner volume ratio output sections 38 and 53 shown in Fig. 2 and Fig. 3.

This allows controlled variable ΔVI to optimum inner volume ratio in the present rotational frequency Hz to be calculated accurately by the arithmetic processing of the optimum inner volume ratio output sections 38 and 53.

Thus, in this embodiment, the electric motor 11 in the screw compressor is inverter-driven by the inverter 15. The position of the slide valve 19 in an axial direction specifying the starting position of discharge is controlled by controlling the working fluid supplied to the operating chamber in the cylinder 24 by the fluid control device 28 based on the control signal from the compression section controller 27.

Regulating of capability to a load is performed by calculating rotational frequency Hz by the revolution number output sections 37 and 52 constituting part of the control devices 36 and 51 to obtain freezing capability required for cold water temperature Tw as load side information and then by controlling so that the revolution number of the electric motor 11 becomes the rotational frequency Hz under the control of the inverter 15 and 54. Therefore,

the need for unload control is eliminated when capability regulation is performed, with the result that the operational efficiency is prevented from lowering. Furthermore, a capacity control valve for capacity control is eliminated, with the result that the valve control mechanism is simplified.

The variable inner volume ratio is obtained by performing arithmetic processing based on the low pressure LP of the suction side, the high pressure HP of the discharge side and the rotational frequency Hz by the optimum inner volume ratio output sections 38 and 53 of the control devices 36 and 51 to calculate optimum inner volume ratio (or ΔVI) in the present rotational frequency Hz, and to determine the starting position of discharge by setting the position of the slide valve 19 in an axial direction by the compression section controller 27 and the fluid control device 28. Therefore, inner volume ratio can be set so that compressor efficiency is maximized corresponding to the rotational frequency Hz of the electric motor 11.

According to this embodiment, when the rotational frequency Hz of the screw compressor 31 is controlled to regulate capability to a load, the decline in compressor efficiency can be minimized.

Fig. 5 shows the relation of freezing capability and compressor efficiency. The horizontal axis indicates freezing capability Q, in which the freezing capability of a conventional variable inner volume ratio-mode screw compressor using variable inner volume ratio and unload control together at a frequency of 60 Hz is expressed as 100% in percentage. On the other hand, the vertical axis indicates compressor efficiency. The above compression ratio varies from 2.1, 3.9, 5.5 to 7.9.

According to the figure, in the case of the variable inner volume ratio-mode inverter screw compressor which employs the variable inner volume ratio and the inverter control of this embodiment together, compressor efficiency is improved at any compression ratio in the freezing capability Q equal to or lower than 100% as compared to the case of the

conventional variable inner volume ratio-mode screw compressor which employs variable inner volume ratio and unload control together. In addition, as freezing capability is lower, compressor efficiency become more improved, with the result that a larger effect is gained. In the case of this variable inner volume ratio-mode inverter screw compressor, capability regulation to a load is controlled by the inverter. Therefore, capability regulation equal to or higher than 100% can be performed. It is to be noted that a conventional screw compressor, which performs capability regulation to a load by unload control, cannot naturally perform capability regulation equal to or higher than 100%.

In the case of a screw compressor, the difference of internal pressures is generated by rotational frequency even under the identical pressure condition, which means that each optimum inner volume ratio value exists corresponding to each frequency. Figs. 6A and 6B show the relation of inner volume and pressure in the case at a frequency of 30 Hz (Fig. 6A) and the case at a frequency of 90 Hz (Fig. 6B). The broken lines in the figures are curved lines which show the relation of inner volume and pressure in the case of the fixed inner volume ratio in which inner volume ratio is fixed to the optimum inner volume ratio value for a frequency of 60 Hz. It is to be noted that the dashed lines are curved lines which show the relation of inner volume and pressure during theoretical adiabatic compression.

In the fixed inner volume ratio, when a frequency is 30 Hz, short of compression is generated at Point (A) so that the pressure is drastically reduced. When a frequency is 90 Hz, excess of compression is generated at Point (B) so that the pressure is substantially increased compared to theoretical value. Thus, an inverter is not simply applicable to capacity control of a screw compressor.

However, as in this embodiment, short of compression generated when inner volume ratio is fixed in the case of a frequency of 30 Hz is not caused and the pressure variation range is narrowed by making inner volume ratio variable as shown in full line. Also, excess of compression generated when inner volume ratio is fixed in the case of a

frequency of 90 Hz can be solved and the pressure range can be narrowed.

It is to be noted that this embodiment has described based on the case in which the capability/inner volume ratio control system of this variable inner volume ratio-mode inverter screw compressor is applied to a refrigerator, but the present invention is not limited to this case. The point is that it is sufficient for the detection signal which is inputted into the revolution number output sections 37 and 52 of the control devices 36 and 51 in Fig. 2 and Fig. 3, to indicate the state of a load.